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International Compressor Engineering Conference

School of Mechanical Engineering

1998

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HIGH-SPEED, DIRECT-DRIVE CENTRIFUGAL COMPRESSORS FOR COMMERCIAL HVAC SYSTEMS

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ABSTRACT

This paper describes progress being made on the development of small capacity(e.g., 10-100 tons) direct drive, oil-free centrifugal compressors. The technology development required for successful application involves high speed motors and drives, centrifugal aerodynamics, oil-less bearings, controls, system modeling, and heat exchangers operating with environmentally safe HFC refrigerants. A Consortium has been formed for the purpose of developing and demonstrating this concept. This project, partially funded by NIST and NYSERDA, is in its third and final year. After successful completion in 1996 of individual component assessments and demonstrations and 1997 proof-of-concept compressor testing at UTRC, another proof-of-concept prototype compressor is now being tested in actual system applications in the Syracuse development laboratories of Carrier.

INTRODUCTION

One of the many challenges facing the HVAC industry is the development of reliable, cost effective, highly efficient, compact, and quiet compression technologies to serve the air/water-cooled commercial marketplace from approximately 5 to 100 tons. An opportunity exists, in this period of transition for compression and refrigerant technology, for the development of small capacity (e.g., 5 to 100 tons) centrifugal compressor technology. Small centrifugals offer significant potential advantages in efficiency (15-20% at design point), size/weight (30-50% reduction), and part count(30-50% fewer) over competing reciprocating, screw, and scroll compression technology.

The technology development required for successful application will involve high speed motors and drives, centrifugal aerodynamics, oil-less bearings, controls, system modeling, and heat exchangers operating with environmentally safe HFC refrigerants. Analytical modeling and experimental demonstrations to date have been encouraging and have proven the technical feasibility of the concept. This includes demonstrations of;

- High-speed, direct-drive, refrigerant cooled motors
- High efficiency (mid to upper 90%) variable speed drives
- Operational testing of cost effective, refrigerant lubricated ball bearings
- High efficiency (mid 80% isentropic) centrifugal aerodynamic performance across the required operating envelope.

High-speed centrifugal compressors have previously been used in some military aircraft environmental control systems[1,2] where weight is at a premium and cost targets are less stringent than for Commercial HVAC systems. Small capacity centrifugal compressor research and development for aircraft applications[3] continues, as well as other HVAC applications such as automotive air conditioning[4,5]. United Technologies Corporation has a long history of centrifugal compressor technical innovation, including numerous projects on small capacity , high-speed, centrifugal compressors[6,7].

The emphasis for this effort is on the development and demonstration of the component technology necessary in order to develop an efficient, cost effective, and reliable compressor for Commercial HVAC systems. The continuing performance improvements and cost reductions in variable speed drives makes this concept increasingly commercially viable.

APPLICATION

The target applications include both air and water cooled chillers, as well as rooftop units. After assessing a number of refrigerants, mechanical configurations, motor and drive options, and candidate demonstration platforms, a 25 ton machine designed for HFC-134a was selected for the proof-of-concept. The compressor was optimized for a rooftop unit operating with a 95F(35C) outdoor design point temperature and 45/130F(7/54C) saturated suction and condensing temperatures. This gives a design point pressure ratio of about 4:1, although enough margin was built in to allow operation at higher ambient temperatures where pressure ratio's approach 4.5:1. Based on the predicted operating range and efficiency, this design should also provide good performance for water cooled systems operating at a nominal 36/103F(2/39C) saturated suction and condensing temperatures. This pressure ratio is a little over 3:1.

AERO CONCEPT SELECTION

A two-stage compressor design was selected instead of a single-stage design because of the lower speed requirement, the increase in impeller size for the same specific speed, avoidance of high tip Mach numbers, and the thrust balancing benefit. At the same time, the two-stage compressor concept allows the use of an economizer refrigeration cycle. This offers the potential of boosting cycle efficiency by 5 to 7 percent, depending on actual operating conditions. Two types of two-stage designs were considered: the in-line two-stage design and the back to back design (see Figure 1).

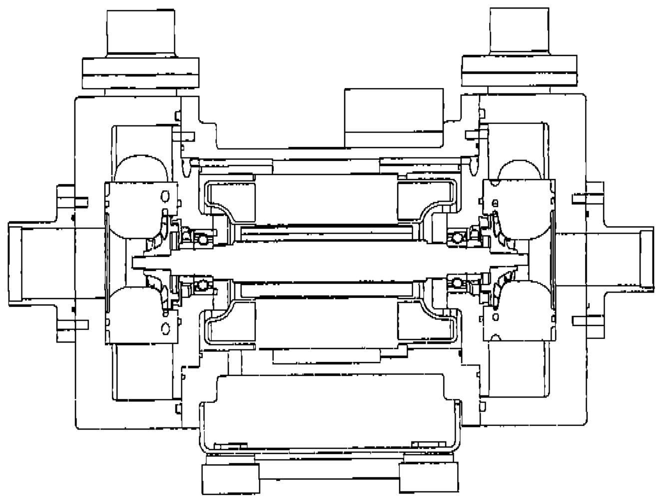

Figure 1 - Two-Stage, Back-to-Back Configuration, Centrifugal Compressor Cross-Section

The in-line design is more compact but it results in lower compressor efficiency due to the relatively sharp turns interstage flow undergoes in the return channel. As a result of these turning losses, the aerodynamic stage efficiency of an in-line two-stage machine is a number of points lower than the stage efficiency of a well-designed single stage machine. In practice, the economizer advantage of a two-stage machine is often negated by the lower cycle efficiency of the individual stages.

In a back-to-back two-stage compressor design, each stage can be designed as a single stage machine with corresponding high stage efficiency. In this case the cycle advantage of the economizer is not compromised by lower individual stage efficiency.

The impeller has splitter blades to increase the impeller throat area and reduce the loading in the exducer. The design uses highly backswept blades (Figure 2). This increases the stable operating range which is important for variable speed operation and is known to result in the highest compressor efficiency.

For small capacity applications, the advantage of improved aspect ratio of the gas passage becomes an important advantage of the high backsweep design.

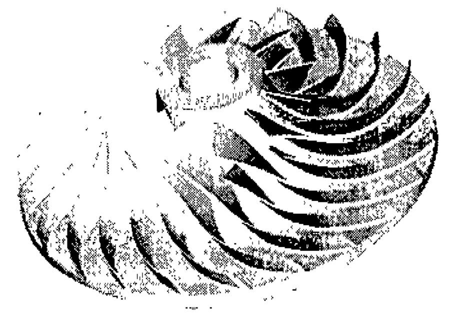

Figure 2 - Prototype Impeller Design

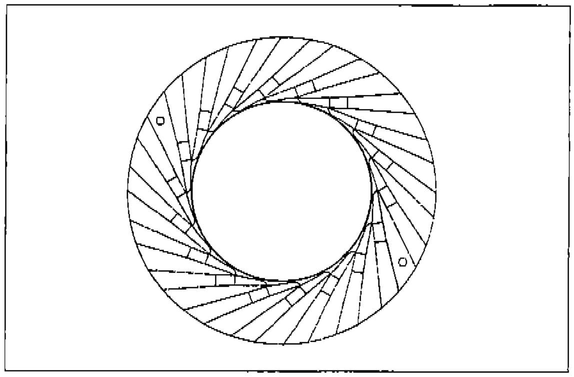

Figure 3 - Prototype Pipe Diffuser Cross-Section

In order to achieve maximum stage efficiency the pipe diffuser concept (Figure 3) that has been applied successfully to large capacity single-stage refrigeration compressors was chosen for each of the two stages. One uncertainty about this design was to what extend Reynolds number effects would reduce the efficiency demonstrated by the high backsweep impeller/pipe diffuser/collector technology at high tonnage centrifugal compressors (200 to 1500 tons). Initial calculations showed an ASME compressor Reynolds number of 500,000, a factor 10 less than the Reynolds number of the large compressors used in large tonnage rotary chillers, but still large for air compressors. Air compressors with this Reynolds number and discrete passage diffusers have reportedly achieved efficiencies well in excess of 80 percent.

Carrier's in-house mean streamline performance prediction program predicted a peak stage efficiency of 84 percent (see Figure 4).

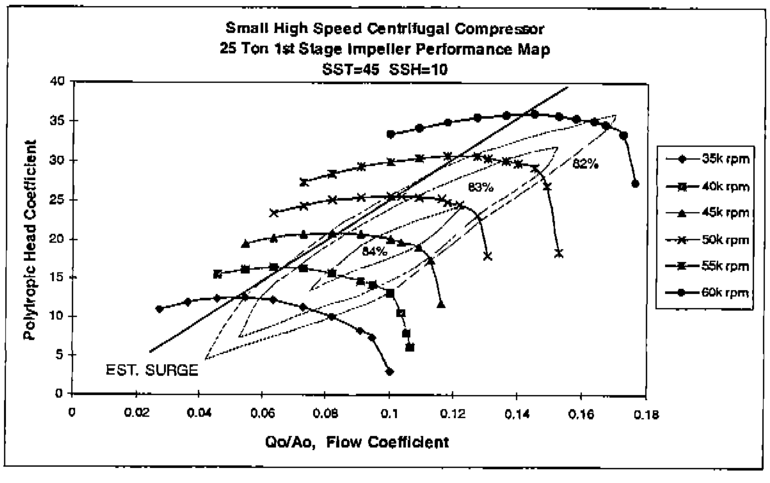

Figure 4 - Predicted 1st Stage Aero Performance

MECHANICAL DESIGN

The proof-of-concept prototype compressor design provides a simple machine with significantly fewer total and moving parts than most positive displacement compresssors. There are fifteen major parts, as shown in Figure 5. The compact 25 ton prototype compressor weighs approximately 85 lbs.(40 kg.) and is approximately 15"(38 cm.) long and 9.5"(24 cm.) in diameter. After allowing for the variable speed drive necessary to run the centrifugal at design speed, this still represents a 30% to 50% weight reduction over other compression technology[8].

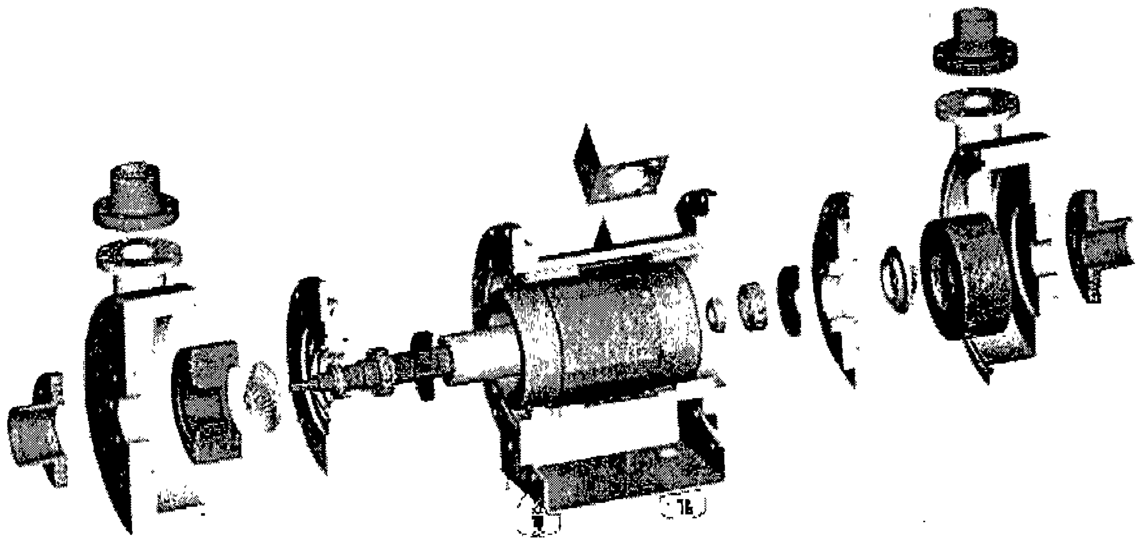

Figure 5 - Twenty Five Ton Prototype Compressor Exploded View

Details on the critical speeds, bearings selection, and motor cooling have previously been reported on[9]. The compressor is oil-free, relying on refrigerant lubricated bearings. Thermal management of the compact, high-speed motor and bearings was a prime consideration in the design. Labyrinth seals to minimize impeller tip leakage as well as leakage from each bearing cavity into the motor housing were carefully designed for the high operating speeds.

Assembly processes have been verified and refined using two different proof-of-concept prototype compressors. These machines have been assembled and disassembled numerous times for inspection and functional evaluation. Initial estimates of fabrication and assembly operations hold promise for relatively small capital investments and cost effective manufacturing if a decision is made to commercialize this technology.

Two motor designs were completed for the prototype. The initial prototype incorporates an induction motor rated for 28 HP at 47,000 RPM. High-speed dynamometer testing indicated an electromagnetic efficiency of 94% at full load, with a relatively flat efficiency across the operating envelope. A second motor design has been constructed and will undergo dynamometer testing shortly. Two different high-frequency output variable speed drives have been designed and fabricated for testing also. Both air and liquid cooled drives will be tested. The initial testing has demonstrated drive efficiency approaching the design goal.

EXPERIMENTAL RESULTS

Shakedown testing of the initial prototype started last year in a desuperheater facility at UTRC. The initial focus was on thermal and mechanical performance. Verification of the bearing lubrication and motor and bearing cooling system performance was accomplished. Bearing and motor winding temperatures have proven very controllable based on lubrication levels and adequate filtering of the lubrication system has been shown to be practical. Many hours of testing over a range of speeds has shown fairly constant compressor efficiency over a wide operating range The emphasis is now shifting to detailed performance testing and component loss distribution evaluation (aero, mechanical, etc.)

Since April 1998, a second prototype unit has been running in the Carrier laboratories in Syracuse as part of an overall HVAC system evaluation. The test platform is a modified water cooled chiller. The test facility is designed such that desired combinations of compressor suction and discharge conditions simulating air or water cooled applications can be generated. This second prototype, operating in a complete system, has duplicated the wide operating range observed in the desuperheater testing. Again the initial focus has been on thermal management and mechanical performance of the compressor. Internal bearing and motor temperatures have proven very responsive to lubrication system variations. Initial test results have confirmed peak aerodynamic polytropic stage efficiencies (total-to-static) approaching the design targets.

One significant challenge has been dealing with electrical noise interference between the proof-of-concept prototype high-frequency variable speed drive / motor combination and the data acquisition system, including pressure transducers, thermocouples, and power analyzers. Careful attention must be paid to this issue as the data acquisition is prone to random noise as well as bias errors.

Figures 6 shows the chiller system component test configuration and instrumentation layout. The system uses brazed plate heat exchangers for the condenser, evaporator, and economizer. Individual control over each bearing lubrication path is facilitated via electronic expansion valves. The compressor is instrumented internally with thermocouples on the bearings, motor windings, rotor-stator gap, cooling jacket, and inlet and outlet of each stage. Pressure measurements are made at inlet and outlet of each stage, as well as each auxiliary refrigerant stream. The intent is to be able to accurately measure overall as well as individual component performance and evaluate opportunities to improve the initial design.

Another interesting potential benefit to this technology is the size and weight implications for the complete HVAC system. For example, the water cooled chiller product that is being used as a test platform could be approximately 30% smaller and weigh approximately 20% less with a high-speed centrifugal compressor.

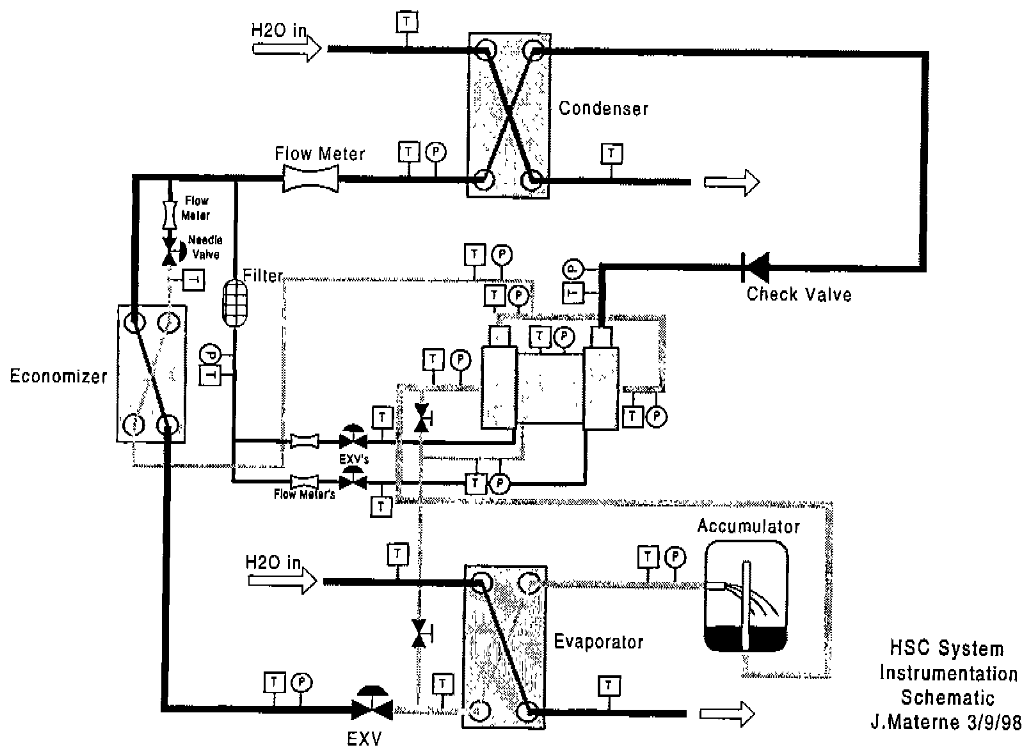

Figure 7 - System Instrumentation

FUTURE PLANS

The emphasis of the testing is shifting away from functionality and toward performance. Analytical assessments on each of the compressor components indicated the potential for 10% to 20% design point advantage over competitive compression technology. The goal for the rest of the program will be to verify these predictions and explore the full extent and compressor performance over the required operating envelope for both air and water cooled Commercial HVAC applications.

ACKNOWLEDGEMENTS

The authors would to acknowledge the funding support and project participation of the National Institute of Standards and Technology's Advanced Technology Program and the New York State Energy Research and Development Authority. The UTC(Carrier and UTRC) project team also appreciates the efforts and contributions of their Consortium partners including General Electric, Lockheed Martin Control Systems, Satcon Technology, Allied Signal, and Dupont. The contributions of Dr. Joost Brasz and Mr. Vishnu Sishtla of Carrier Corporation on the aerodynamic concepts and mechanical design have also significantly benefited the project.

REFERENCES

1. Cloud, , W. W., J. E. McNamara, and D. B. Wigmore, "A New Technology in Energy-Efficient Electrically Driven Aircraft Environmental Control Systems," ACS Paper 869390, presented at the 21st Intersociety Energy Conversion Conference, August, 1986.
2. Springer, T., J. E. McNamara, J. C. Lentz, and D. B. Wigmore, "Testing of an Energy Efficient Environmental Control System for Patrol-Type Aircraft," SAE Paper 921225, presented at the 22nd International Conference on Environmental Systems, July, 1992
3. Gui, F., T. R. Reinarts, and R. P. Scaringe, "Design and Experimental Study of High-Speed, Low-Flow-Rate Centrifugal Compressors," IECEC Paper No. CT-39, ASME, 1995.
4. Peterson, G. E., "A High Efficiency. Low Mass, Modular Automotive Air Conditioning System," Paper C496/056/95, IMechE, 1995.
5. Yun, H., and J. L. Smith, "Centrifugal Compressors for Automotive Air Conditioners - Component Design," AES-Volume 36, Proceedings of the ASME Advanced Energy Systems Division, pp. 115-122, 1996.
6. Biancardi, F., and D. P. Darrow, "A Turbocompressor Air-Conditioning and Heating System for the Home," ASHRAE Annual Meeting, Paper No. 2104, January 1969.
7. Biancardi, F., J. Sitler, and G. Melikian, "Development and Test of Solar Rankine Cycle Heating and Cooling Systems," ASHRAE Transactions. Volume 88, Part 1, 1982.
8. Eames, I. W., D. Howe, R. L. Elder, C. W. R. McFarlane, and G. W. Graham, "A Novel Lightweight Compressor," Paper C477/005/94, pp. 9-16, IMechE, 1994.
9. Pandy, D. R. and D. Brondum, "Innovative, Small, High-Speed Centrifugal Compressor Technologies," presented at the 1996 International Compressor Engineering conference at Purdue, July, 1996.